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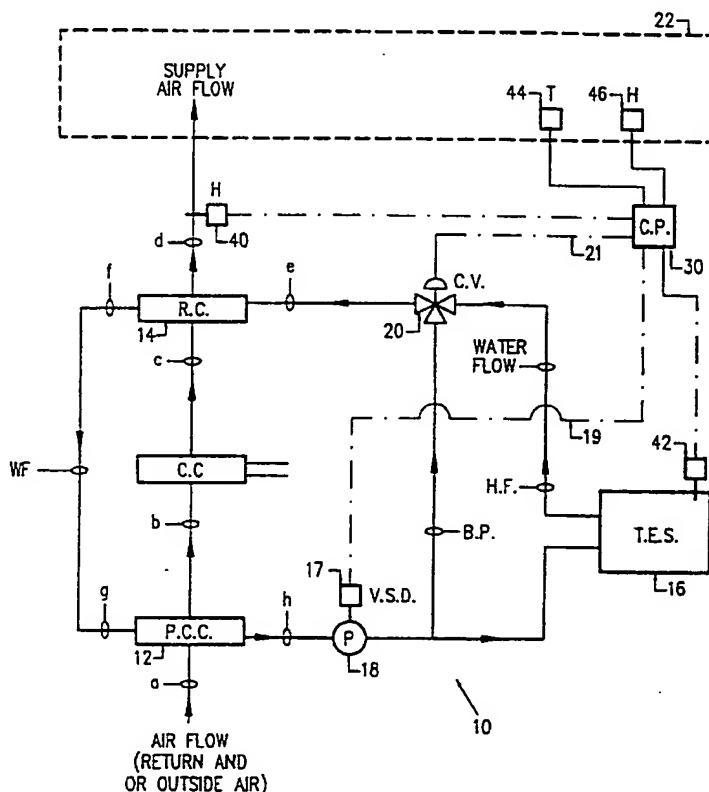
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(54) Title: METHOD AND APPARATUS FOR LATENT HEAT EXTRACTION

(57) Abstract

A method and apparatus for improved latent heat extraction combines a run-around coil system (WF) with a condenser heat recovery system (56) to enhance the moisture removing capability of a conventional vapor compression air conditioning unit (50). The run-around coil system (WF) exchanges energy between the return (a) and supply air (d) flows of the air conditioning unit (10). Energy recovered in the condenser heat recovery system (56) is selectively combined with the run-around system energy extracted from the return air flow (a) to reheat the supply air (d) stream for downstream humidity control. A control system (30) regulates the relative proportions of the extracted return air flow energy and recovered heat energy delivered to the reheat coil (14) for efficient control over moisture in the supply air flow (d). Auxiliary energy in the form of electric heat energy (60) is further added to the recovered heat energy for additional reheat use.



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**METHOD AND APPARATUS
FOR LATENT HEAT EXTRACTION**

Background of the Invention

This application pertains to the art of air conditioning methods and apparatus. More particularly, this application pertains to methods and apparatus for efficient control of the moisture content of an air stream which has undergone a cooling process as by flowing through an air conditioning cooling coil or the like. The invention is specifically applicable to dehumidification of a supply air flow into the occupied space of commercial or residential structures. By means of selective combination of extracted return air flow heat energy and recovered refrigerant waste heat energy, the supply air flow is warmed using a reheat coil apparatus. The return air flow entering the air conditioning coil is precooled with a precooling coil in operative fluid communication with the reheat coil. Heating of the occupied space may be effected using the combined reheat and precooling coils in conjunction with an alternative heat source such as electric, solar, or the like and will be described with particular reference thereto. It will be appreciated, though, that the invention has other and broader applications such as cyclic heating applications wherein a supply air flow is heated at the reheat coil irrespective of the instantaneous operational mode of the refrigerant system through the expedient of a thermal energy storage tank or the like.

Conventional air conditioning systems use a vapor compression refrigeration cycle that operates to cool an indoor air stream through the action of heat transfer as the air stream comes in close contact with evaporator type or flooded coil type refrigerant-to-air heat exchangers or coils. Cooling is accomplished by a

reduction of temperature as an air stream passes through the cooling coil. This process is commonly referred to as sensible heat removal. A corresponding simultaneous reduction in the moisture content of the air stream typically also occurs to some extent and is known as latent heat removal or more generally called dehumidification. Usually the cooling itself is controlled by means of a thermostat or other apparatus in the occupied space which respond to changes in dry bulb temperature. When controlled in this manner, dehumidification occurs as a secondary effect incidental to the cooling process itself. As such, dehumidification of the indoor air occurs only when there is a demand for reduced temperature as dictated by the thermostat.

To accomplish dehumidification when the thermostat does not indicate a need for cooling, a humidistat is often added to actuate the air conditioning unit in order to remove moisture from the cooled air stream as a "byproduct" function of the cooling. In this mode of operation, heat must be selectively added to the cooled air stream to prevent the conditioned space from over-cooling below the dry bulb set point temperature. This practice is commonly known as "reheat".

Many sources of heat have been used for reheat purposes, such as hydronic hot water with various fuel sources, hydronic heat recovery sources, gas heat, hot gas or hot liquid refrigerant heat, and electric heat. Electric heat is most often used because it is usually the least expensive alternative overall. However, the use of electric heat to provide the reheat energy is proscribed by law in some states, including Florida for example.

In order to conserve energy, it has been suggested that recovered heat be used as a source for the reheat. Accordingly, one method to improve the moisture removal capacity of an air conditioning unit, while simultaneously providing reheat, is to provide two heat exchange surfaces each in one of the air streams entering or leaving the cooling coil while circulating a working fluid between the two heat exchangers. This type of simple system is commonly called a run-around system.

Run around systems have met with limited success. The working fluid is cooled in a first heat exchange surface placed in the supply air stream called a reheat coil. The cooled working fluid is then in turn caused to circulate through a second heat exchange surface placed in the return air stream called a precooling coil. This simple closed loop circuit comprises the typical run-around systems available heretofore.

The precooling coil serves to precool the return air flow prior to its entering the air conditioning cooling coil itself. The air conditioning coil then provides more of its cooling capacity for the removal of moisture from the air stream otherwise used for sensible cooling. However, the amount of reheat energy available in this process is approximately equal to the amount of precooling accomplished. This is a serious constraint. Additional reheat energy is often needed for injection into the run-around system to maintain the desired dry bulb set point temperature and humidity level in the conditioned space. As described above, supplemental electric reheat has been used with some success.

In addition, the growth of molds in low velocity air conditioning duct systems has recently become a major indoor air quality concern. One of the control measures recognized as having the capability of limiting this undesirable growth is the maintenance of the relative humidity at 70 percent or lower in the air conditioning system air plenums and ducts. Within limits, reheat can be used to precisely control the relative humidity. However, as described above, the amount of reheat energy from the run-around systems available today may not be sufficient to consistently provide the above level of humidity control, particular during periods of operation when the air temperature entering the precooling coil is lower than the system design operating temperature.

As a further complication, air conditioning units are also often used for heating purposes as well as for cooling and dehumidification. Electric heating elements are often provided in the air conditioning units to selectively provide the desired amount of heat at precise times of the heating demand. The above demand for heating energy will most often correspond with the demand for heating at other air conditioning units in the locality. This places a substantial and noticeable demand on the electrical power utility system in the community. In many areas, this peak demand has exceeded the capacity of the power system. The electric utility companies have responded with incentives encouraging their customers to temper their demand during regional peak demand periods. These incentives are often in the form of demand charges which encourage the customer to reduce their demand on the system at those times in order to avoid incremental costs in addition to the regular base rates.

It has, therefore, been deemed desirable to provide an economical solution that meets the various needs of air conditioning system installation requirements while also operating in compliance with current and projected local environmental and energy-related laws.

Summary of the Invention

This invention improves the dehumidification capabilities of conventional air conditioning systems through the addition of a run-around system having a supplemental heat energy source for reheat use. The amount of reheat energy that can be incrementally added to the stream air leaving the conditioning unit is thereby increased. An air conditioning unit so configured is capable of operating continuously over a wide range of conditions for providing dehumidification to the occupied space independent of the sensible cooling demand at the conditioned space. Such a system is further capable of maintaining a precise relative humidity level in the air conditioning duct system to enhance the indoor air quality of the occupied conditioned space. Further, the overall system may be used to heat the occupied space through the expedient of the stored energy scheme according to the teachings of the preferred embodiments.

In the preferred embodiment, the supplemental heat source is heat recovered from the refrigeration process of the particular installed air conditioning system having the reheat requirement. In another embodiment, the supplemental heat is an alternative energy source, such as a gas or electric boiler, or water heater. The new energy source may be of

particular benefit for use with an air conditioning system that uses chilled water or cold brine for the cooling medium.

5 The basic preferred embodiment of the invention comprises heat exchange coils in the entering air stream and leaving air stream of an air conditioning unit primary cooling coil. The basic preferred embodiment further comprises a circulating pump, and a
10 supplementary heat source, which can be a heat recovery device on the air conditioning unit refrigeration circuit or a conventional liquid heater or the like.

Brief Description of the Drawings

15 FIGURE 1 illustrates a schematic view of the preferred embodiment of the apparatus for latent heat extraction according to the invention;

FIGURE 2 illustrates a schematic view of the preferred embodiment of the invention when used with a conventional air conditioning unit having a vapor
20 compression type refrigeration system;

FIGURE 3 illustrates a schematic of the preferred embodiment of the invention when used with an air conditioning unit using chilled water for the cooling medium;

25 FIGURES 4a, 4b are flow charts of the control procedure executed by the control apparatus during the space cooling mode of operation;

FIGURES 5a, 5b are flow charts of the control procedure executed by the control apparatus during the
30 space dehumidification mode of operation;

FIGURE 6 is a flow chart of the control procedure executed by the control apparatus during the space heating mode of operation;

FIGURE 7 is a flow chart of the control procedure executed by the control apparatus during the various operational modes for maintenance of the thermal energy storage tank temperature;

5 FIGURE 8 is a coil graph of a first sample calculation;

FIGURE 9 is a coil graph of a second sample calculation;

10 FIGURE 10 is a coil graph of a third sample calculation; and,

FIGURE 11a, 11b are a psychometric chart of the combined first, second and third sample calculations and a protractor for use with the psychometric chart.

15 Detailed Description of the Preferred Embodiments

Referring now to the drawings wherein showings are for purposes of illustrating the preferred embodiments of the invention only and not for purposes of limiting same, the FIGURES show a moisture control apparatus 10 for conditioning the air in an occupied space 22. The apparatus 10 comprises suitably arranged components including a precooling coil 12 in a return air flow a,b, a reheat coil 14 in a supply air flow c,d, a thermal energy storage tank 16 operatively associated with a source of heat, a working fluid pump 18 for circulating a working fluid WF through a series arrangement of the above coils, a variable speed drive 17 for controlling the speed of pump 18 and a modulated control valve 20 for metering the working fluid. An apparatus controller 30 directly modulates the control valve 20 and generates variable speed command signals for control over the working fluid pump 18.

With particular reference to FIGURE 1, the working fluid WF enters the control valve 20 from one of

two sources including a bypass fluid flow BP and a heated fluid flow HF, the latter passing first through the thermal energy storage tank 16. In both above cases, the flow of the working fluid is motivated by the working fluid pump 18. A mixture of bypass fluid flow BP and heated fluid flow HF may be accomplished over a continuum by a blending control valve substituted for the modulated control valve 20, along with an analog output signal from the apparatus controller 30 described below.

The apparatus controller 30 is an operative communication with a plurality of system input devices, each of which sense various physical environmental conditions. These input devices include a supply airflow humidity sensor 40, a thermal energy storage tank temperature sensor 42, an occupied space dry bulb temperature sensor 44, and an occupied space humidity sensor 46. The humidity sensor 40 may be replaced with a temperature sensor for ease of maintenance and reliability.

In addition, the controller 30 is in operative communication with a plurality of active output devices. The output devices are responsive to signals deriving from the apparatus controller 30 according to programmed control procedures detailed below. In the preferred embodiment, the output devices comprise the control valve 20 responsive to a control valve signal 21, and a variable speed drive 17 responsive to a pump speed command signal 19. Additional input and output signals, including alarm and data logging signals or the like, may be added to the basic system illustrated in FIGURE 1 as understood by one skilled in the art after reading and understanding the instant detailed description of the preferred embodiments.

With particular reference now to FIGURE 2, a schematic diagram of the preferred embodiment of the apparatus of the invention is illustrated adapted for use with a conventional air-conditioning unit having a vapor compression type refrigeration system. The system includes a compressor 50 for compressing a compressible fluid CF and a condenser coil 52. An evaporative cooling coil 54 absorbs heat from a return air flow a, b resulting in a cooled supply air flow c, d into an occupied space 22. These various air conditioning components may be assembled in a single package, known in the art as a roof-top unit, or may be provided as a system comprising separated items, such as what is called a split system.

With continued reference to FIGURE 2, a reheat coil 14, as described above, is placed in the supply air flow c, d after (downstream of) the evaporative cooling coil 54, while a precooling coil 12 is placed in the return air flow a, b before (upstream of) the cooling coil 54. For full effectiveness of the air quality control measure of the instant invention, the reheat coil 14 should be physically mounted as close as possible to the cooling coil 54. The precooling coil 12 can be mounted in any convenient location and may be so situated as to precool only the outside air, only the return air, or a mixture of the outside air and return air (not shown).

As above, the working fluid pump 18 is connected to a variable speed drive 17 which operates to circulate the working fluid WF between the reheat coil 14, the precooling coil 12, and the thermal energy storage tank 16. In the preferred embodiment, the working fluid is water. In general, the overall system may be used in various operating modes including a space

cooling mode, a space dehumidification mode, and a space heating mode. To describe the full operation of the system, each of the operational modes will be described in detail below.

5 In the space cooling mode, the working fluid pump 18 operates when the refrigeration system compressor 50 is operating. In this mode, the compressor 50 is responsive to the occupied space dry bulb temperature sensor 44. The pump 18 is driven by
10 the variable speed drive 17 which regulates the water flow to maintain the desired humidity setting at the supply air flow humidity sensor 40. Water flow is increased on a rise in the relative humidity above a predetermined set point and conversely decreased on a
15 drop in relative humidity at the supply air flow humidity sensor 40 below said set point.

 In the space dehumidification mode, the compressor 50 of the conventional air-conditioning unit is operated to maintain the humidity at the occupied
20 space 22, as sensed by the occupied space humidity sensor 46, the speed of the working fluid pump 18 is regulated to maintain the desired temperature of the occupied space 22 as sensed by the occupied space dry bulb temperature sensor 44. In this dehumidification
25 mode of operation, working fluid flow WF is increased on a drop in temperature at the occupied space dry bulb temperature sensor 44, and water flow is conversely decreased on a rise in the occupied space temperature. Responsive to command signals from the apparatus
30 controller 30 and according to the control algorithms detailed below. When the temperature in the occupied space is a controlling factor in setting the working fluid pump speed, the supply air flow humidity set point is used to establish at a minimum working fluid pump

speed. In any of the above modes, working fluid flow control may be accomplished using a two-port valve with a modulating actuator in place of the variable speed drive 17.

5 In general terms, cooled air leaving the evaporative type cooling coil 54 enters the reheat coil 14 where it absorbs heat from the working fluid flow in the tubes of the reheat coil itself. There is a drop in heat content of the working fluid from points e to f
10 equal to the rise in the heat content of the air stream from points c to d. The working fluid is transferred through the piping system 32 to the precooling coil 12. Cooled working fluid from the reheat coil 14 absorbs
15 heat from the return air flow stream as the air passes over the precooling coil surfaces. There is a rise in the heat content in the working fluid from points g to h equal to the drop in the heat content of the air stream from points a to b. These principles are each generally well-known and established in the art.

20 Heat exchange pump 58 operates when the compressor 50 is operating and when the temperature and the thermal energy storage tank 16 is below a predetermined set point at the thermal energy storage tank temperature sensor 42. The function of the heat
25 exchange pump 58 is to transfer working fluid heated by the hot refrigerant gas in a heat exchanger 56. The heat exchange pump 58 stops even though the compressor
30 50 is running when the temperature in the thermal energy storage tank 16 is at an upper working fluid temperature set point as determined by the thermal energy storage tank temperature sensor 42. The general function of the heat exchanger 56 is to provide supplemental heat to charge the thermal energy storage tank 16 with hot working fluid for heating and/or reheat operation.

An electric heating element 60 may be used as an additional energy source to heat the working fluid when there is a demand for more heat than may be provided by the heat exchanger 56. The supplemental electric heating operation is controlled by the apparatus controller 30 to operate as a secondary source of energy when the temperature in the thermal energy storage tank 16 drops below the desired set point as determined by the thermal energy storage tank temperature sensor 42. As an example, if the desired minimum temperature in the thermal energy storage tank is 120°F and the desired maximum temperature is 125°F, the heat exchange pump 58 is made to begin operation on a drop in temperature below 120°F. Conversely, when the thermal energy storage tank temperature drops to 120°F, the electric heating element 60 is activated by the apparatus controller 30. On a rise in the thermal energy storage tank temperature, the heating element 60 is first turned off, and on a continued rise in temperature to the 125°F set point, the heat exchange pump 58 is next turned off. This scheme is hierarchically arranged in order to conserve energy by first recovering energy from the air-conditioning unit which might otherwise be lost.

Multiple heating elements similar to the electric heating element shown may be provided and controlled by a step controller to match the energy input to the heating load in stages of electric heat. An SCR controller may be used to proportionally control the amount of heat energy added to the thermal energy storage tank 16 as a function of the tank temperature differential from minimum to maximum set points. On a larger scale, such as neighborhood-wide, the electric heating controls may be circuited to allow the lock-out

of the electric heating elements during periods of peak electrical demand throughout the neighborhood. This lock-out control may be in the form of an external signal, such as may be provided from the neighborhood power company, or from the owner's energy management system. The control may further be obtained from a signal from the system controls contained in the apparatus controller 30, as a function of the time of day, demand limiting, or other energy management strategies.

Referring next to FIGURE 3, a schematic diagram of the preferred embodiment of the invention is illustrated and modified for use with an air-conditioning unit using chilled water as the cooling medium. The chilled water system uses a chilled water cooling coil 70 which may be mounted in a duct or plenum, or can be mounted in an air-handling unit with integral or remote mounted fans. Chilled water systems are usually provided with a control valve 72 to regulate the amount of cooling accomplished by the system in response to the occupied space dry bulb temperature sensor 44.

With continued reference to FIGURE 3, a reheat coil 14, as described above, is placed in the supply air flow c,d after the evaporative cooling coil 54, while a precooling coil 12 is placed in the return air flow a,b before the cooling coil 54. For full effectiveness of the air quality control measure of the instant invention, the reheat coil 14 should be mounted as close as possible to the cooling coil 54. The precooling coil 12 can be mounted in any convenient location and may be so situated as to precool only the outside air, only the return air, or a mixture of the outside air and return air (not shown).

The pump 18 is connected to a variable speed drive 17 which operates to circulate the working fluid WF, in this preferred embodiment water, between the reheat coil 14, the precooling coil 12, and the thermal energy storage tank 16. In general, the overall system may be used in various operating modes including a space cooling mode, a space dehumidification mode, and a space heating mode. To describe the operation of the system, each of the operational modes will be introduced here and described in detail below.

In the space cooling mode, the working fluid pump 18 operates when there is a demand for cooling in space 22. In this mode, the control valve 72 is responsive to the occupied space dry bulb temperature sensor 44. The pump 18 is driven by the variable speed drive 17 which regulates the water flow to maintain the desired humidity setting at the supply air flow humidity sensor 40. Water flow is increased on a rise in the relative humidity above a predetermined set point and conversely decreased on a drop in relative humidity at the supply air flow humidity sensor 40 below said set point.

In the space dehumidification mode, the air-conditioning unit is operated to maintain the humidity at the occupied space 22, as sensed by the occupied space humidity sensor 46, the speed of the working fluid pump 18 is regulated to maintain the desired temperature of the occupied space 22 as sensed by the occupied space dry bulb temperature sensor 44. In this dehumidification mode of operation, working fluid flow WF is increased on a drop in temperature at the occupied space dry bulb temperature sensor 44, and water flow is conversely decreased on a rise in the occupied space temperature. Responsive to command signals from the

apparatus controller 30 and according to the control algorithms detailed below. When the temperature in the occupied space is a controlling factor in setting the working fluid pump speed, the supply air flow humidity set point is used to establish at a minimum working fluid pump speed. In any of the above modes, working fluid flow control may be accomplished using a two-port valve with a modulating actuator in place of the variable speed drive 17.

In general terms, cooled air leaving the type cooling coil 70 enters the reheat coil 14 where it absorbs heat from the working fluid flow in the tubes of the reheat coil itself. There is a drop in heat content of the working fluid from points e to f equal to the rise in the heat content of the air stream from points c to d. The working fluid is transferred through the piping system 32 to the precooling coil 12. Cooled working fluid from the reheat coil 14 absorbs heat from the return air flow stream as it passes over the precooling coil surfaces. There is a rise in the heat content in the working fluid from points g to h equal to the drop in the heat content of the air stream from points a to b. These principles are each generally well-known and established in the air.

An electric heating element (not shown) may be used as a supplemental energy source to heat the working fluid when there is a demand for additional heat. The supplemental electric heating operation is controlled by the apparatus controller 30 to operate as a secondary source of energy when the temperature in the thermal energy storage tank 16 drops below the desired set point as determined by the thermal energy storage tank temperature sensor 42. As an example, if the desired minimum temperature in the thermal energy storage tank

is 120°F and the desired maximum temperature is 125°F, the electric heating element (not shown) is activated by the apparatus controller 30 when the thermal energy storage tank temperature drops to 120°F. On a return in
5 the thermal energy storage tank temperature to 125°F, power to the heating element is turned off.

Multiple heating elements similar to the electric heating element described above may be provided and controlled by a step controller to match the energy
10 input to the heating load in stages of electric heat. An SCR controller may be used to proportionally control the amount of heat energy added to the thermal energy storage tank 16 as a function of the tank temperature differential from minimum to maximum set points. On a
15 larger scale, such as neighborhood-wide, the electric heating controls may be circuited to allow the lock-out of the electric heating elements during periods of peak electrical demand throughout the neighborhood. This lock-out control may be in the form of an external
20 signal, such as may be provided from the neighborhood power company, or from the owner's energy management system. The control may further be obtained from a signal from the system controls contained in the apparatus controller 30, as a function of the time of
25 day, demand limiting, or other energy management strategies.

With reference now to FIGURES 2, 3, 4a and 4b, the control method for the space cooling mode operation will be described. In the space cooling mode, the
30 compressor 50 of FIGURE 2 and the chilled water cooling coil 70 of FIGURE 3 are operated 104, 106 to maintain the desired set point dry bulb temperature in the occupied space 22 according to the occupied space dry bulb temperature sensor 44. In the conventional air-

conditioning system, the compressor 50 starts 106 on a rise in occupied space temperature above a predetermined set point and stops 104 on a fall in occupied space temperature below the set point temperature 102 as sensed by the occupied spaced dry bulb temperature sensor 44. Correspondingly, in the chilled water system, the control valve 20 opens 106 on a rise in the occupied space temperature and closes 104 on a fall in the occupied space temperature below the predetermined set point at occupied space dry bulb temperature sensor 44. In either case, the speed of the working fluid pump 18 is regulated by the variable speed drive 17 to maintain the desired relative humidity 110 in the supply air flow d as sensed by the supply air flow humidity sensor 40.

The pump speed is also controlled to maintain the desired relative humidity 108 in the occupied space 22 according to the occupied space humidity sensor 46. The working fluid pump speed increases 114 on a rise in the relative humidity above the supply air or the occupied space air relative humidity set points. The working fluid pump speed decreases 112 on a fall in the relative humidity below the set points.

When the variable speed drive 17 is at full speed 118, the control valve 20 is modulated to maintain the desired humidity set points 120, 122. The control valve 20 is positioned to bypass the thermal energy storage tank 16 when the working fluid pump 18 is operating at speeds of less than 100% of full speed. When the variable speed pump 18 is at full speed, the control valve 20 is modulated open 126 to thermal energy storage tank 16 on a rise in supply air 122 or occupied space 120 relative humidity above the predetermined set points according to the supply air flow humidity sensor

40 and the occupied space humidity sensor 46 respectively. In this state, the working fluid flows to the reheat coil 14 directly from the thermal energy storage tank 16 as a heated working fluid flow HF. The control valve 20 is modulated closed 124 on a decrease in the supply air or occupied space air relative humidity below the predetermined set points.

Next, with reference to FIGURES 2, 3, 5a and 5b, the control method for the space dehumidification operating mode will now be described. During this mode, when the occupied space dry bulb temperature set point is satisfied according to the occupied space dry bulb temperature sensor 44, the compressor 50 of the conventional air conditioning unit is operated to maintain the desired occupied space relative humidity. In the chilled water system, the chilled water control valve 72 is operated to maintain the desired occupied space relative humidity. In this mode, the compressor 50 or the chilled water control valve 72 operate 208 on a rise in the occupied space relative humidity 202 above the set point and stop 206 on a drop in the occupied space relative humidity 202 below said set point. The working fluid pump 18 and control valve 20 are controlled 210-222 according to the space cooling mode described above.

With reference next to FIGURES 2, 3 and 6, the control method for the space heating operating mode will now be described. In this mode, the thermal energy storage tank 16 is utilized to maintain the desired occupied space dry bulb temperature according to the physical conditions sensed by the occupied space humidity sensor 46. Normally in this mode, the compressor 50 and chilled water control valve 72 are both off in the standard air-conditioning system and

chilled water systems respectively. In the instant space heating mode, the working fluid WF is circulated exclusively through the thermal energy storage tank 16 as a heated fluid flow HF. No flow is permitted through the bypass as a bypass fluid flow BP. This is accomplished via the control valve 20 modulated open 302 according to the control valve signal 21 from the apparatus controller 30. The speed of the working fluid pump 18 is adjusted 306, 308 to maintain the desired temperature set point 304 in the occupied space 22. As an alternative means, the working fluid pump 18 may be continuously operated, but cycled on and off according to the demand for heating as sensed by the occupied space dry bulb temperature sensor 44. This results in an average heating defined by the duty cycle of the alternating on/off cycles.

With reference now to FIGURE 7, the thermal energy storage tank maintenance routine TES will be now described in detail. The method is a subroutine in each of the space cooling, space dehumidification, and space heating control methods described above. In this control subroutine procedure, heat exchange pump 58 operates 408 when the compressor 50 is operating 402 and when the temperature in the thermal energy storage tank 16 is below the set point 404 at temperature sensor 42. The function of pump 58 is to transfer water WF heated by the hot refrigerant gas in the heat exchanger 56. The pump stops 406 when the temperature in the tank is at the upper water temperature set point 404 at the temperature sensor 42. The function of the heat exchanger is to provide supplemental heat to charge the thermal storage tank 16 with hot water for heating and/or reheat operation.

Electric heating element 60 may be used as an additional energy source to heat the water when there is a demand for more heat than can be provided by the heat exchanger. The electric heating operation is controlled by the apparatus controller 30 to operate 414 as the second source of energy when the temperature in the thermal storage tank 16 drops below the desired set point 410 at sensor 42. As an example, if the desired minimum temperature in the tank is 120° F and the desired maximum temperature is 125° F, the pump 58 starts on a drop in temperature below 125° F. When the tank temperature drops to 120° F, the electric heating element 60 is activated. On a rise in tank temperature the heating elements are turned off first 416, and on a continued rise in temperature to 125° F the pump 58 is, in turn, shut off 406. Multiple heating elements may be provided and controlled by a step controller to match the energy input to the heating load in stages of electric heat or an SCR controller can be used to proportionately control the amount of heat energy added to the tank as a function of the tank temperature differential from minimum to maximum set points.

The electric heating controls may further be circuited to allow for a lock out 416 of the electric heating elements during periods of peak community electrical demand 412. This lock out control could be provided from an external signal such from the power company or from the owner's energy management system. The control could be from a signal from the system controls contained in control 30 as a function of time of day, demand limiting, or other energy management strategies.

With reference once again to FIGURE 2, the system may be operated in a variety of modes. In

general, when the overall system is operating in either the cooling mode or the dehumidifying mode the cold air leaving the evaporator coil 50 enters the reheat coil 14 where it absorbs heat from the moving water stream WF in the tubes of the reheat coil 12. There is a corresponding drop in the heat content of the circulating water from points e to f equal to the rise in heat content of the air stream from points c to d. The water WF is transferred through a piping conduit system to the precooling coil. Cold water entering the precooling coil 12 absorbs heat from the return air stream a as it passes over the coil surfaces. There is a rise in heat content of the circulating water from points g to h equal to the drop in heat content of the air stream from points a to b. Representative sample calculations follow below.

SAMPLE CALCULATIONS

The sample calculation A immediately below is illustrated in the coil graph of FIGURE 8 and in the psychometric chart of FIGURES 11a, 11b wherein it is Given that:

- Required indoor temperature is 75°F at 45% relative humidity;
- Indoor cooling load (peak load) is
220.0 MBTU/Hour Sensible
94.3 MBTU/Hour Latent
314.3 MBTU/Hour Total;
- Outdoor air temperature at peak cooling load is 93°F dry bulb and 76° dry wet bulb;
- Amount of ventilation air (outside air) required is 2500 CFM;
- Desired supply air relative humidity level is 70% maximum;

- Return air heat gain assumed equal to a 2°F ΔT rise; and
- Fan and motor heat gain assumed equal to a 1½°F ΔT rise.

5 Statement of Solution:

- Sensible heat ratio = $\frac{220.0}{314.3} = 0.70$
- Room condition line intersects 70% RH line at 55°F.
- 10 - Supply air volume required:

$$V = \frac{220000 \text{ BTU/HR}}{1.1 \cdot 20^\circ \Delta T}$$
- Reheat energy required to provide 70% Rel. Hum. in supply air stream:
15
$$Q = 10000 \text{ CFM} \cdot 1.1 \cdot [(55-47)^\circ \text{F} \Delta T - 1\frac{1}{2}^\circ \text{F}]$$

$$= 71500 \text{ BTU/HR}$$
- Water flow rate required through reheat coil assuming 6½°F ΔT and 12°F approach temperature:
20
$$V = 71500 \text{ BTU/Hour} / (500 \cdot 6.5^\circ \text{F} \Delta T) = 22 \text{ GPM}$$
- Coil conditions - Temperature:

	Air	Water
Entering Coil	47	65.5
Leaving Coil	53.5	59.0
- 25 - Precooling coil air temperature drop (sensible cooling):

$$\Delta T = \frac{Q}{1.1 \cdot \text{CFM}}$$

Q = Amount of energy recovered for supply air stream at reheat coil

$\Delta T = 71500 \text{ BTU/Hour} / 1.1 \cdot 10000 \text{ CFM} = 6.5^\circ \text{F} \Delta T$
- Coil conditions - Temperature

	Air	Water
Entering Coil	81	59
Leaving Coil	74.5	65.5

The sample calculation B immediately below is illustrated in the coil graph of FIGURE 9 and in the psychometric chart of FIGURES 11a, 11b wherein it is
 5 Given that:

- Same condition as calculation (A), except indoor sensible cooling load is 110.0 MBTU/Hour; and,
- Assume supply air dew point is fixed
 10 at 45°F due to coil characteristics;

Statement of Solution:

-New sensible heat ratio

$$\frac{110.0}{110 + 94.3} = 0.54$$

- Reheat energy required
 15 $Q = 10000 \text{ CFM} \cdot 1.1 \cdot [(65-47)^\circ\text{F} \Delta T - 1\frac{1}{2}^\circ\text{F}]$
 $= 181500 \text{ BTU/hour}$

- Water temperature required using 22 GPM flow rate

$$\Delta T = \frac{181500 \text{ BTU/hour}}{22 \text{ GPM} \cdot 500} = 16.5^\circ \Delta T ^\circ\text{F}$$

- Reheat energy required from refrigerant heat recovery:

$$Q_3 = Q_1 - Q_2$$

$$Q_1 = \text{Total reheat required}$$

$$Q_2 = \text{Water heat gain in precooling coil (from Calculation (A))}$$

$$Q_3 = 181500 - 71500 \text{ BTU/hour} = 110,000 \text{ BTU/hour}$$

- Temperature rise required by water through heat reclaim device:
 30

$$\Delta T = \frac{Q_3}{500} \cdot 22 \text{ GPM} = \frac{110000}{500} \cdot 22 \text{ GPM} = 10^\circ\text{F}$$

The sample calculation C immediately below is illustrated in the coil graph of FIGURE 10 and in the psychometric chart of FIGURES 11a, 11b wherein it is Given that:

- 5 - Same conditions as Calculation (A),
 except:
- Space sensible cooling load is 110 MBTU/hour
- Refrigeration compressor(s) provided with
10 capacity reduction to reduce amount of
 refrigerant flow, matching the new cooling
 load; this results in an increased dew point
 in the supply air.

Statement of Solution:

- 15 - Assuming capacity reduction raises
 the supply air dew point to 51°F;
- Space condition line intersects dew
 point line as 65°F db, this is the
 supply air dry bulb temperature;
 space condition line extends up and
20 to the right, establishing a new
 room condition of 75°F at ~ 53%
 relative humidity.

The sample calculation immediately below illustrates the Heating Mode of operation wherein it is Given:

- Space heating load is 216000
BTU/Hour, peak;
- Supply air volume is 10,000 CFM
(from Calculation (A));
- Desired space temperature is 72°F;
- Outside air temperature is 35°F;
- and,
- Outside air volume is 2500 CFM.

Statement of Solution:

- Supply air temperature required is

$$T_s = \frac{72^\circ\text{F} \cdot 216000 \text{ BTU/hour}}{1.1 \cdot 10000 \text{ CFM}} = 72^\circ\text{F} + 20 = 92^\circ\text{F}$$
- Mixed air temperature is:

$$T_m = 72^\circ\text{F} \left[\frac{216000 \text{ BTU/hour}}{10000 \text{ CFM}} \cdot (72 - 35)^\circ\text{F} \right]$$

$$= 62.75^\circ\text{F}$$
- Total heating required

$$Q = 1.1 \cdot 10000 \text{ CFM} \cdot (92 - 62.75)^\circ\text{F}$$

$$= 321750 \text{ BTU/hour} = 94\text{KW}$$
- Heat provided from thermal storage -
 ASSUMPTIONS: full heating shift to
 OFF peak, 10 hour heating period,
 60% diversity.
- Heating required:

$$Q = 10 \text{ hours} \cdot 321750 \text{ BTU/hour} \cdot .6 \text{ diversity}$$

$$= 1930500 \text{ BTU}$$
- Heat input to thermal storage:
 - During moderate temperature periods recovered heat would be used to charge the storage tank. During cold weather, when the cooling system

is off, the electric heat would be used to store the energy.

- Electric heater size:

$$Q = 1930500 \text{ BTU}/14 \text{ hours} = 137900 \text{ BTU/hour}$$

$$= 40 \text{ KW*}$$

- Thermal storage volume required -

ASSUMPTIONS: minimum useful temperature is 100°F and storage temperature is 140°F.

$$V = \frac{1930500 \text{ BTU}}{8.35 \text{ lb/gal.} \cdot 1 \text{ BTU/lb.}^\circ\text{F} \cdot (140 - 100)^\circ\text{F}}$$

$$V = 5780 \text{ Gallons}$$

- The amount of storage could be reduced if the electric heat is allowed to operate during the peak period (at a reduced rate to provide some demand saving):

$$V = \frac{1930500 \text{ BTU} - 10 \text{ hrs} \cdot 20 \text{ KW} \cdot 3413 \text{ BTU/KW}}{8.35 \text{ lb/gal.} \cdot 1 \text{ BTU/lb.}^\circ\text{F} \cdot (140 - 100)^\circ\text{F}}$$

$$V = 3736 \text{ Gallons}$$

* Heater size and/or storage volume would be increased slightly to account for system losses.

The invention has been described with reference to the preferred embodiments. Obviously modifications and alterations will occur to others upon a reading and understanding of this specification. It is my intention to include all such modifications and alterations insofar as they come within the scope of the appended claims and equivalents thereof.

Having thus described the invention, I now claim:

1. A moisture control apparatus for use with a fluid compression air conditioning system having a compressor for compressing a compressible fluid, and a cooling coil where the compressible fluid decompresses absorbing thermal energy from a return air flow as a cooled supply air flow, the moisture control apparatus comprising:

a working fluid;
precooling coil means in said return air flow for exchanging thermal energy between the return air flow and the working fluid;
reheat coil means in said supply air flow for exchanging thermal energy between the working fluid and the supply air flow;
heat exchange means for exchanging thermal energy between the compressible fluid and the working fluid;
fluid pump means for motivating a flow of the working fluid through said precooling coil means, said reheat coil means, and said heat exchange means; and,
regulating means for regulating said working fluid flow through said precooling and reheat coil.

2. A moisture control apparatus according to claim 1 further comprising fluid conduit means containing the working fluid therein for containedly directing the working fluid through a series arrangement of said precooling coil means, said heat exchange means, and said reheat coil means.

3. A moisture control apparatus according to claim 2 wherein said regulating means comprises a control valve connected to said fluid conduit means in said series arrangement.

4. A moisture control apparatus according to claim 3 further comprising bypass conduit means, connected to said control valve in parallel with said heat exchange means and in parallel with a series
5 combination of said precooling coil means and said reheat coil means, for selectively circulating a first portion of the working fluid as a bypass flow through said series combination of said precooling coil means and said reheat coil means.

5. A moisture control apparatus according to claim 4 wherein said control valve comprises a first input port connected to said fluid conduit means for receiving a first flow of said working fluid from said heat exchange means, a second input port connected to
5 said bypass conduit means for receiving said bypass flow of said working fluid from said precooling coil means, an output port connected to said fluid conduit means for selectively exhausting said first and bypass flows from
10 said control valve to said reheat coil means, and valving means for selectively metering said first and bypass flows through said control valve as a metered flow.

6. A moisture control apparatus according to claim 5 wherein said fluid pump means comprises a variable speed drive fluid pump.

7. A moisture control apparatus according to claim 6 further comprising control means operatively associated with said control valve and said variable speed drive fluid pump for sensing moisture in said supply air flow and for maintaining said sensed moisture at a predetermined set point by regulating i) said valving means to selectively meter said first and bypass flows, and ii) said variable speed drive fluid pump to motivate the metered flow through said series combination of said reheat coil means and said precooling coil means.

8. A moisture control apparatus according to claim 7 further comprising thermal energy storage unit means, connected to said fluid conduit means and operatively associated with said working fluid and said heat exchange means, for recovering and storing thermal energy from said compressible fluid and selectively delivering the stored thermal energy to said working fluid.

9. A humidity control apparatus for use with a cooling coil means for selectively absorbing thermal energy from a return airstream as a cooled airstream, the apparatus comprising:

precooling coil means disposed upstream of said cooling coil means for selectively precooling said return airstream;

reheat coil means disposed downstream of said cooling coil means for selectively reheating the cooled airstream flowing from said cooling coil means;

sensing means disposed downstream of said reheat coil means for sensing the relative humidity of the reheated airstream flowing from said reheat coil

- means and generating a humidity signal reflective of
15 said sensed relative humidity;
conduit means connecting said precooling and
reheat coil means for communicating a working fluid flow
through a closed loop circuit comprising said precooling
coil means and said reheat coil means;
20 pump means for controlledely motivating the
working fluid flow through said closed loop circuit at a
controlled flow rate responsive to said humidity signal;
and,
means for selectively introducing thermal
25 energy into said working fluid flow responsive to i)
said humidity signal and ii) said pump means motivating
the working fluid flow at a predefined maximum
controlled flow rate.

10. The humidity control apparatus according to
claim 9 wherein said means for selectively introducing
thermal energy into said working fluid flow comprises:
thermal energy storage means for storing
5 thermal energy;
means disposed downstream of said pump means
and in said closed loop circuit for dividing said
working fluid flow into at least two partial parallel
fluid flows comprising i) a bypass fluid flow and ii) a
10 heated fluid flow passing through said thermal energy
storage means; and,
metering means disposed in said closed loop
circuit upstream of said reheat coil means for receiving
said bypass fluid flow at a first inlet port and said
15 heated fluid flow at a second inlet port and selectively
metering the received fluid flows for exhaust at an
exhaust port connected to said conduit means.

11. A method of heating a supply airstream into a conditioned space from a return airstream and downstream of a cooling coil of a typical air conditioning system operating in a space dehumidification mode for
5 dehumidifying the conditioned space responsive to a humidity sensor in the conditioned space, the method comprising the steps of:

storing heat energy in a thermal energy storage tank;

10 sensing the temperature of the conditioned space;

selectively circulating a working fluid at a controlled flow rate from a precooling coil in the return airstream of the air conditioning system directly
15 to a reheat coil in said supply airstream when said sensed temperature is below a first predetermined set point; and,

selectively circulating the working fluid from the precooling coil through the thermal energy storage
20 tank and the reheat coil in series combination when said controlled flow rate is at a predetermined maximum rate and said sensed temperature is below the first predetermined set point.

12. The method according to claim 11 further comprising the steps of recovering waste thermal energy from said air conditioning system and storing the recovered waste thermal energy in said thermal energy
5 storage tank.

13. A method of controlling the relative humidity of a supply airstream into a conditioned space from a return airstream and downstream of a cooling coil of a typical air conditioning system operating in a space

5 cooling mode for cooling the conditioned space
responsive to a dry bulb temperature sensor in the
conditioned space, the method comprising the steps of:
 storing heat energy in a thermal energy
storage tank;
10 sensing the relative humidity of the supply
airstream;
 selectively circulating a working fluid at a
controlled flow rate from a precooling coil in the
return airstream of the air conditioning system directly
15 to a reheat coil in said supply airstream when said
sensed relative humidity is above a first predetermined
set point; and,
 selectively circulating the working fluid from
the precooling coil through the thermal energy storage
20 tank and the reheat coil in series combination when said
controlled flow rate is at a predetermined maximum rate
and said sensed relative humidity is above the first
predetermined set point.

14. The method according to claim 13 further
comprising the steps of recovering waste thermal energy
from said air conditioning system and storing the
recovered waste thermal energy in said thermal energy
5 storage tank.

15. An apparatus for controlling the relative
humidity of an airstream supplied into a temperature
conditioned space downstream of an air conditioning
cooling coil, the apparatus comprising:
5 control means for controlling the apparatus
according to a predetermined control method;
 precooling means in said airstream for
precooling the airstream upstream of said cooling coil;

reheating means in said airstream for
10 reheating the airstream downstream of said cooling coil;
connecting means for connecting said
precooling means and said reheating means in a series
closed loop;

humidity sensing means in the airstream
15 downstream of the reheating means and connected to said
control means for i) sensing the relative humidity of
the airstream between said reheating means and said
temperature conditioned space and ii) generating a
humidity signal representing the sensed humidity;

20 working fluid pump means responsive to said
control means for selectively pumping a working fluid
through said series closed loop when said humidity
signal is at a predetermined level, the working fluid
pump means having an inherent maximum pumping rate;

25 thermal energy storage tank means in fluid
communication with said connecting means for storing
thermal energy therein; and,

regulating means connected to said connecting
means and said thermal energy storage tank means
30 responsive to said control means for selectively
regulating a flow of said working fluid through said
series closed loop and said thermal energy storage tank
means when said working fluid pump means is pumping at
said maximum pumping rate and said humidity signal is at
35 said predetermined level.

16. An apparatus for controlling the temperature
of an airstream supplied into a moisture conditioned
space downstream of an air conditioning cooling coil,
the apparatus comprising:

5 control means for controlling the apparatus
according to a predetermined control method;

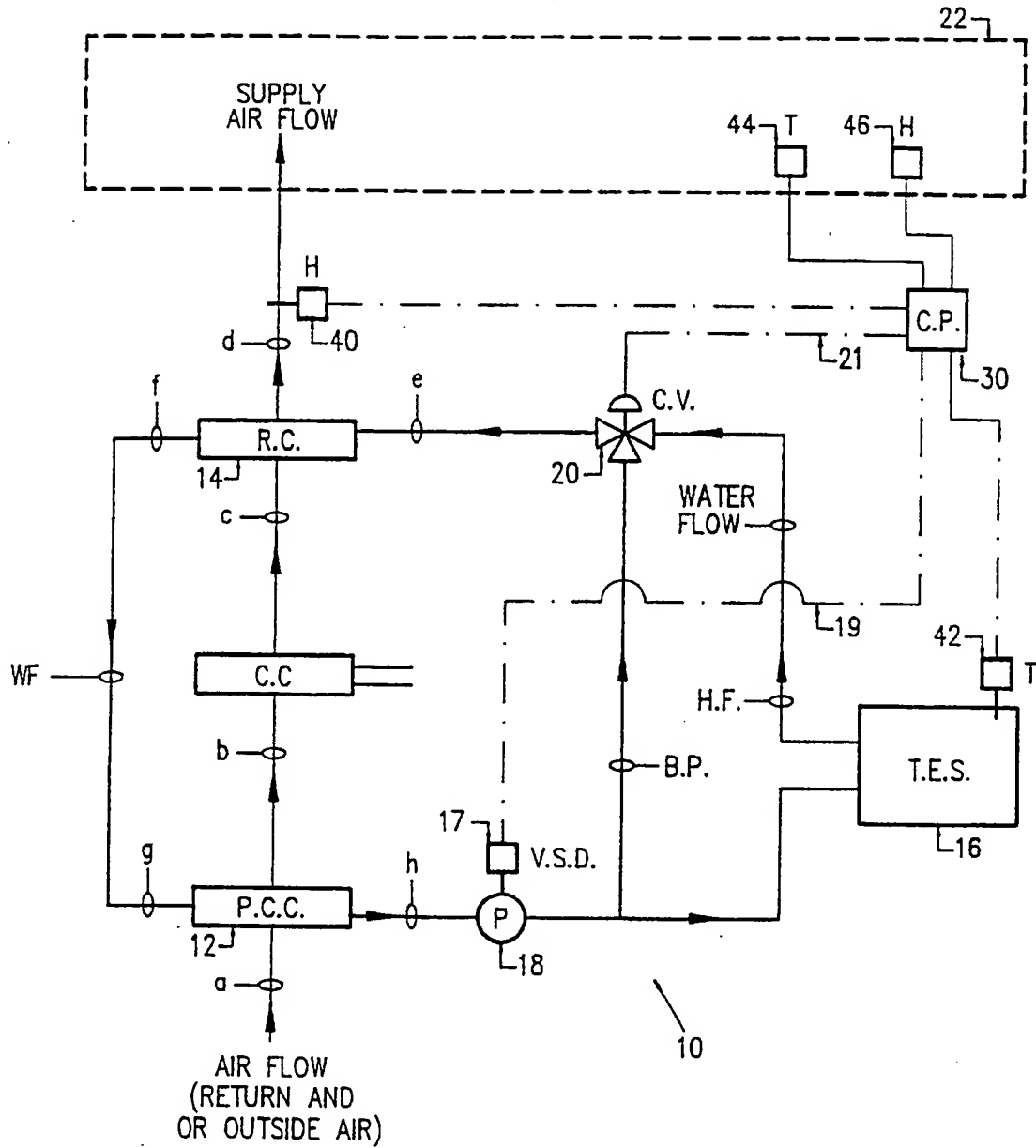
precooling means in said airstream for
precooling the airstream upstream of said cooling coil;
reheating means in said airstream for
10 reheating the airstream downstream of said cooling coil;
connecting means for connecting said
precooling means and said reheating means in a series
closed loop;

15 temperature sensing means in the conditioned
space downstream of the reheating means and connected to
said control means for i) sensing the temperature of the
conditioned space downstream of said reheating means and
ii) generating a temperature signal representing the
sensed temperature;

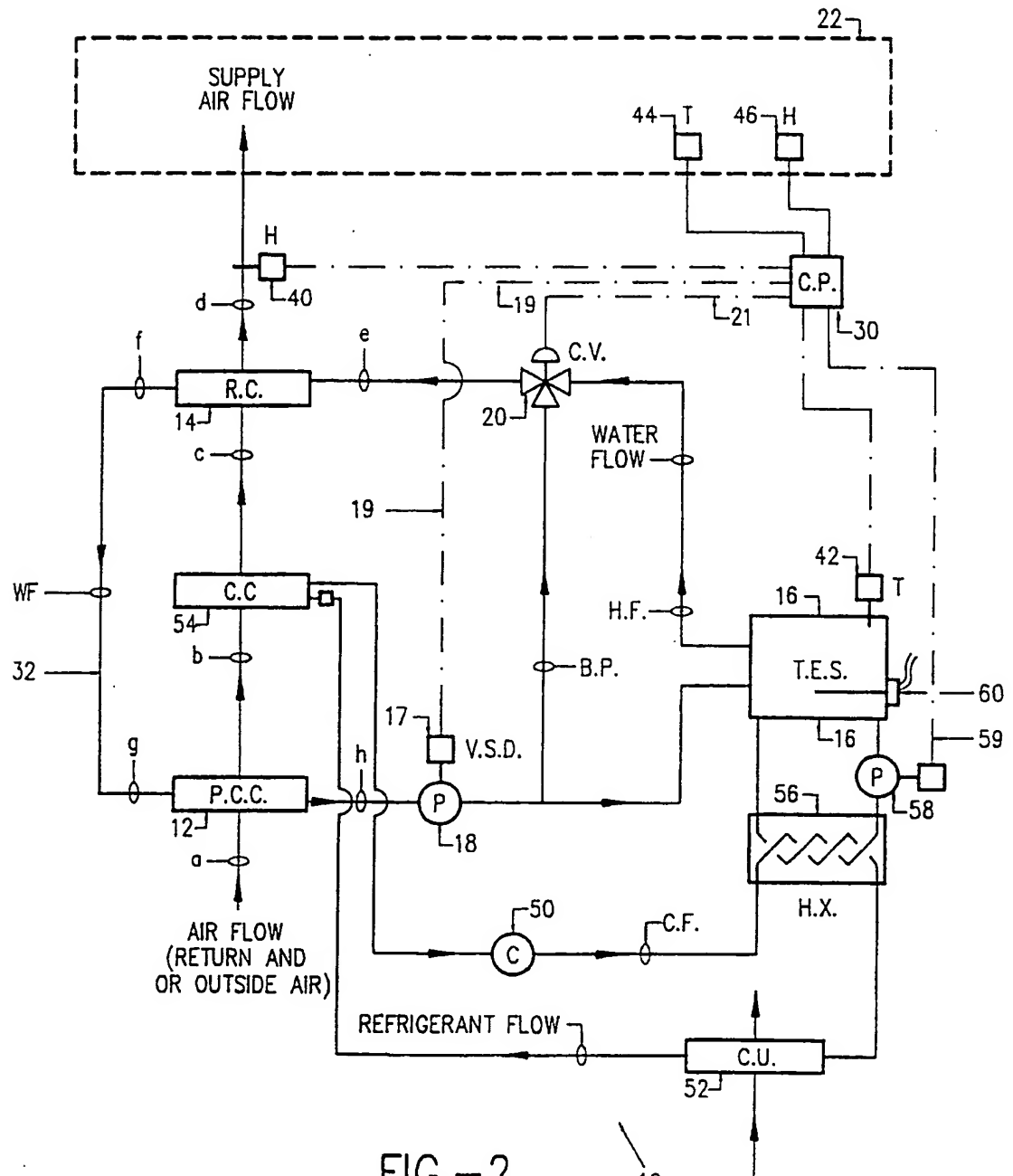
20 working fluid pump means responsive to said
control means for selectively pumping a working fluid
through said series closed loop when said temperature
signal is at a predetermined level, the working fluid
pump means having an inherent maximum pumping rate;

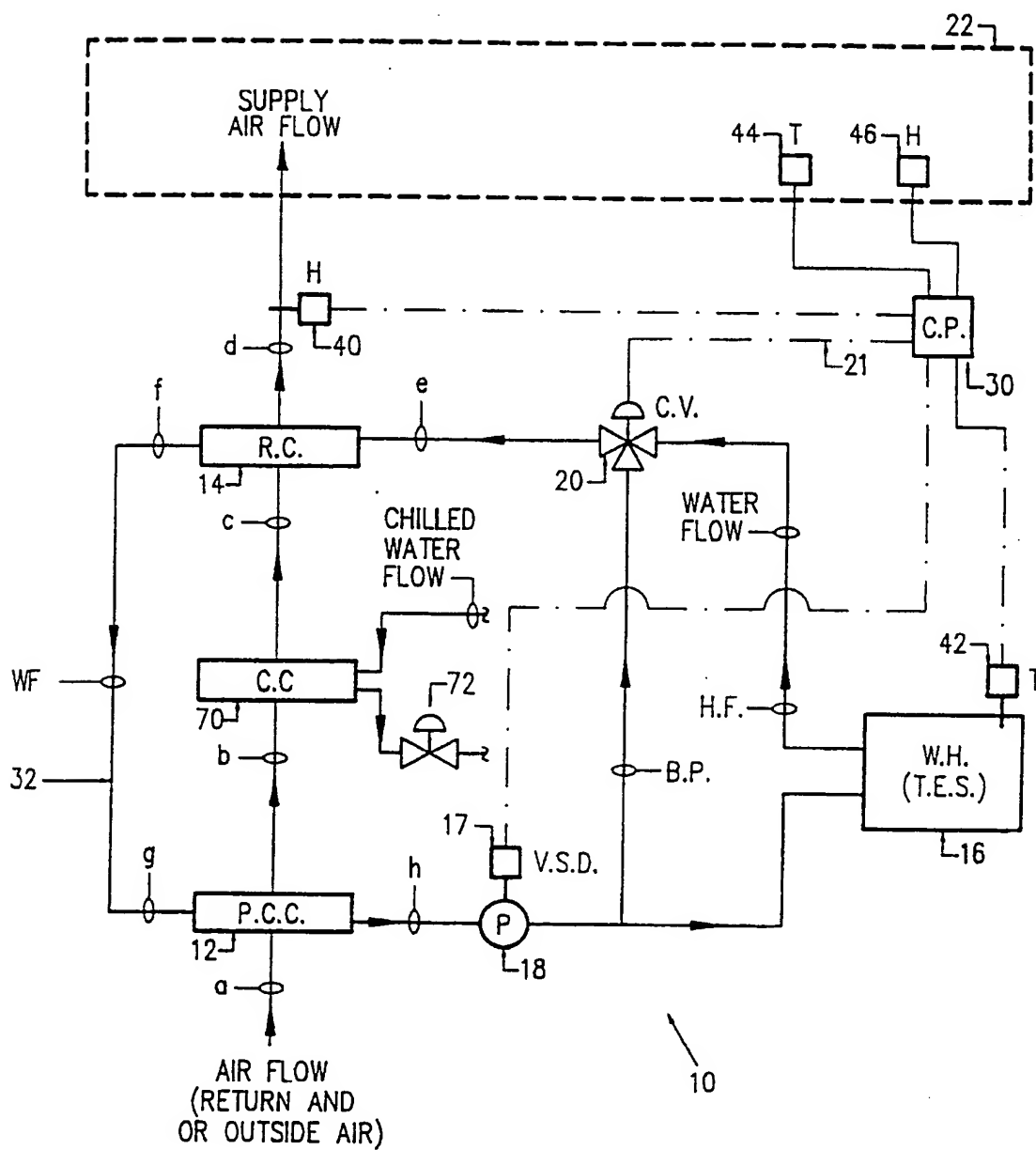
25 thermal energy storage tank means in fluid
communication with said connecting means for storing
thermal energy therein; and,

regulating means connected to said connecting
means and said thermal energy storage tank means
30 responsive to said control means for selectively
regulating a flow of said working fluid through said
series closed loop and said thermal energy storage tank
means when said working fluid pump means is pumping at
said maximum pumping rate and said temperature signal is
35 at said predetermined level.

FIG.-1

2 / 1 4



FIG.-3

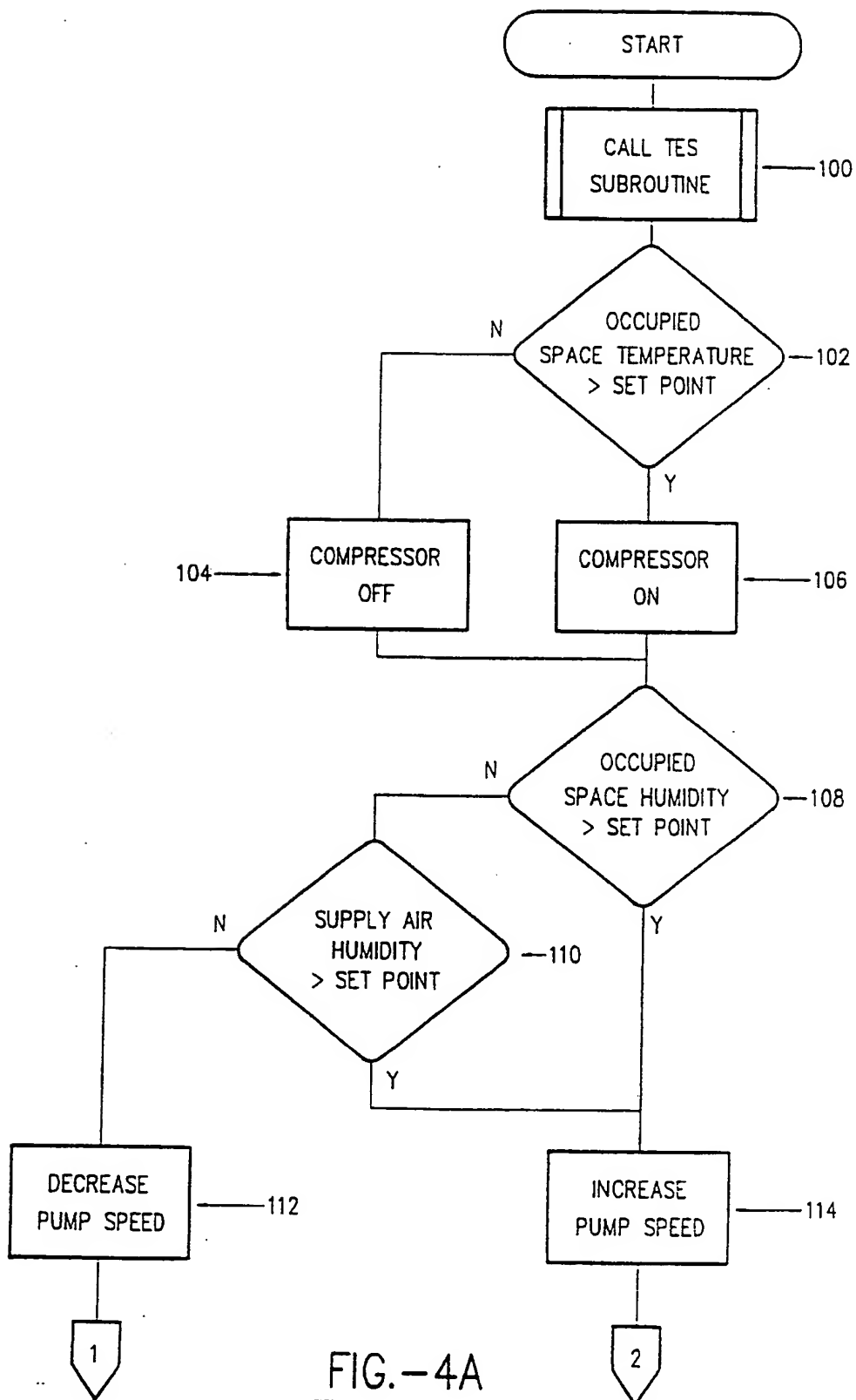
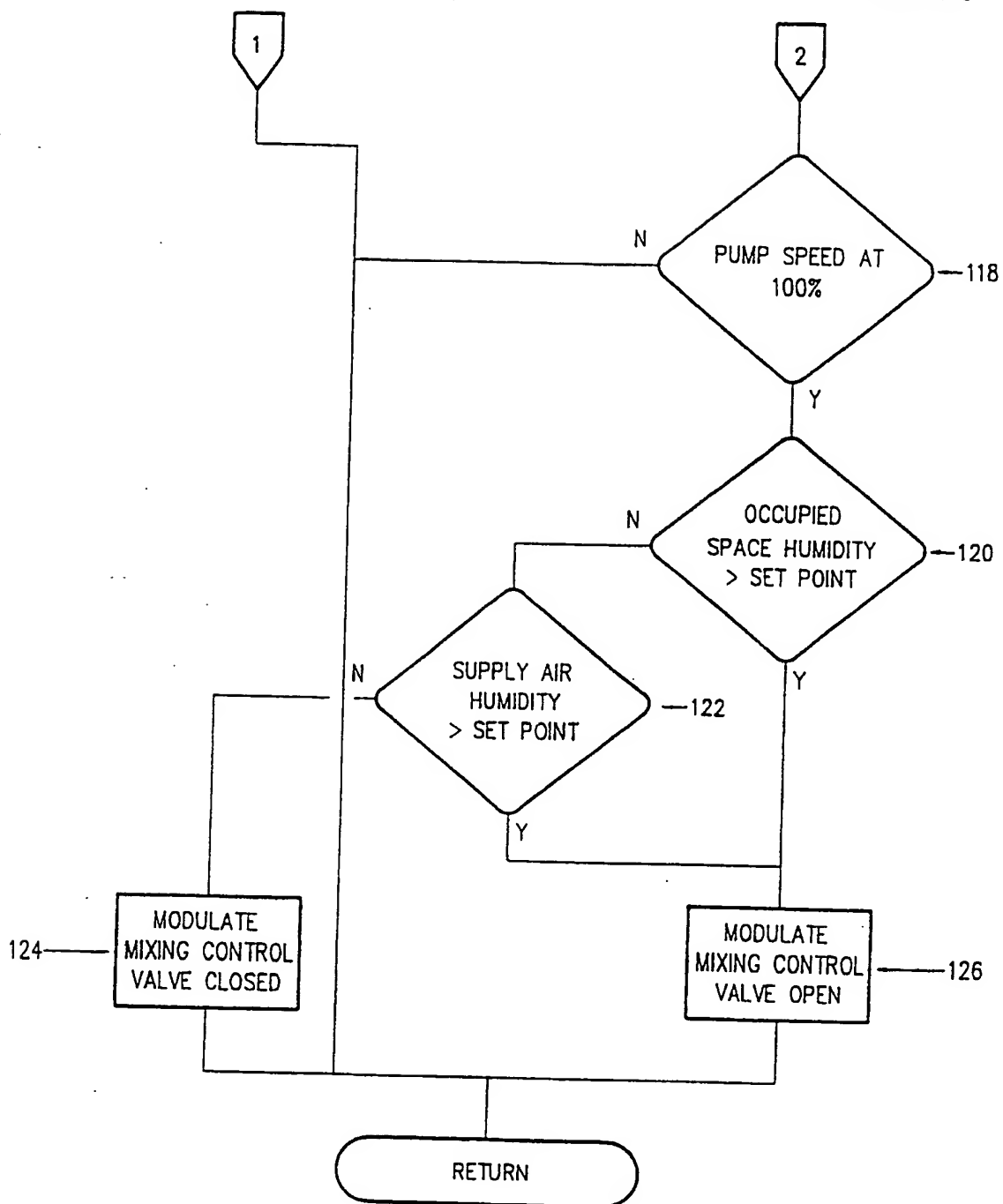
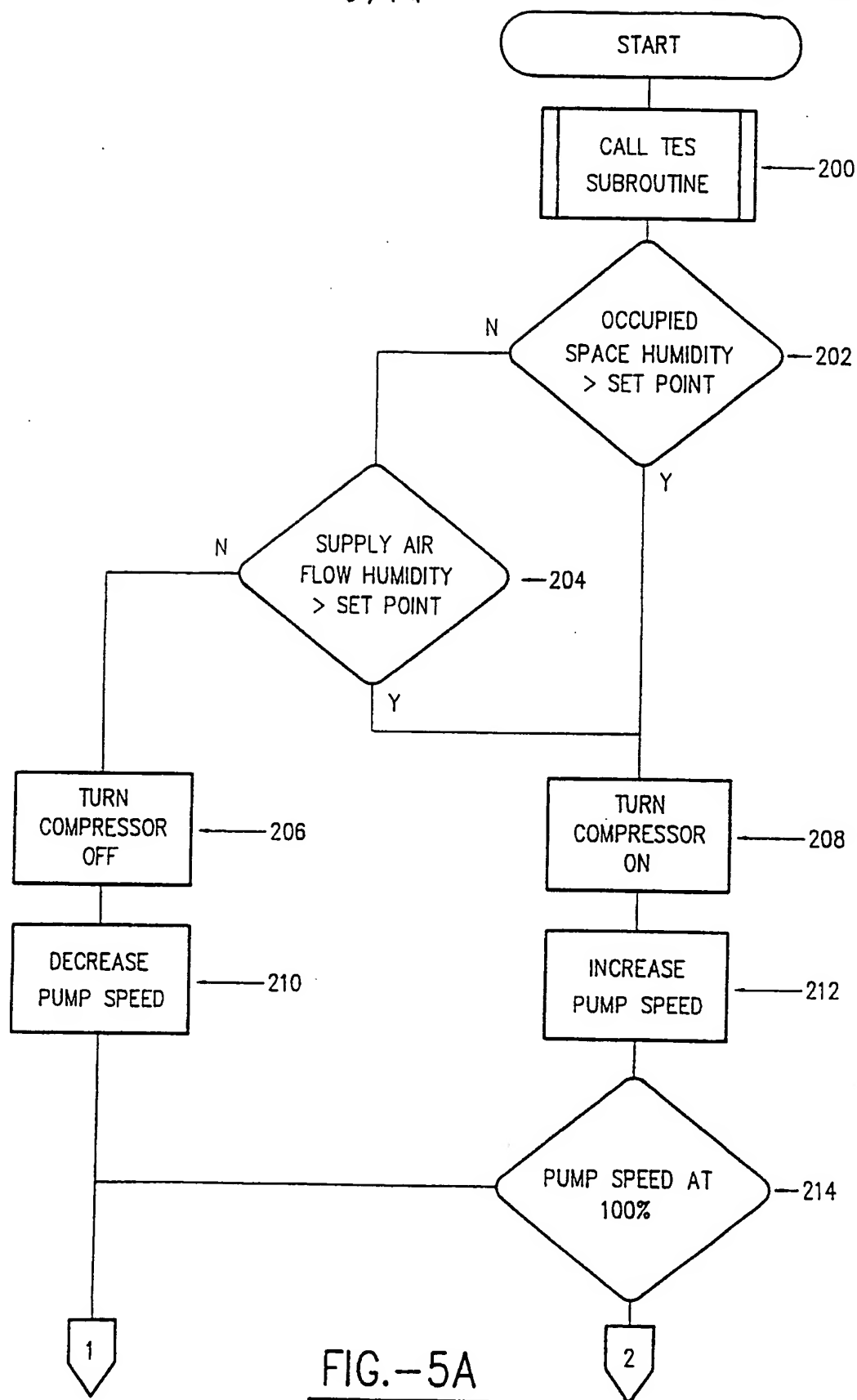
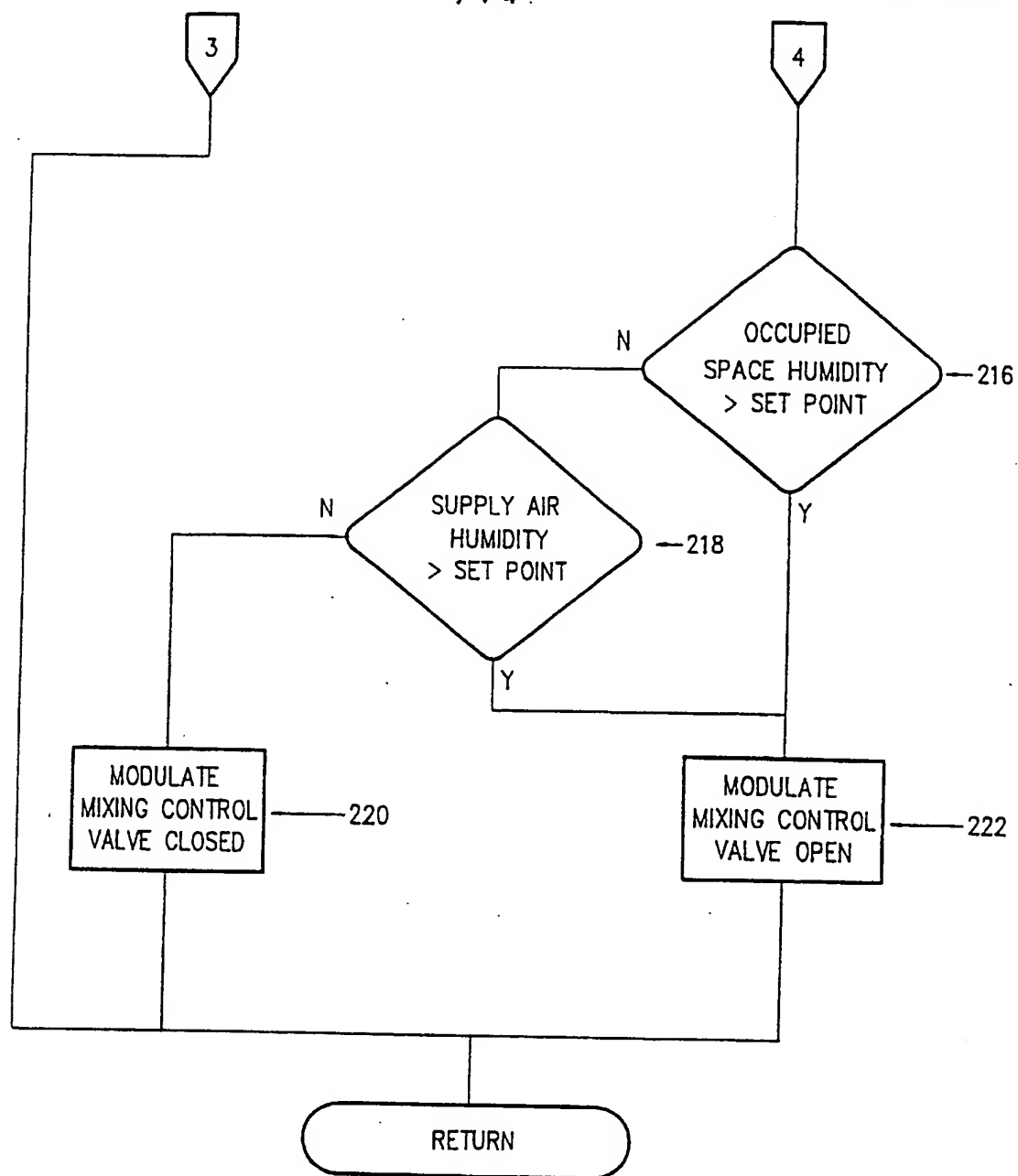


FIG.-4A

FIG. - 4B

FIG. -5A

FIG. - 5B

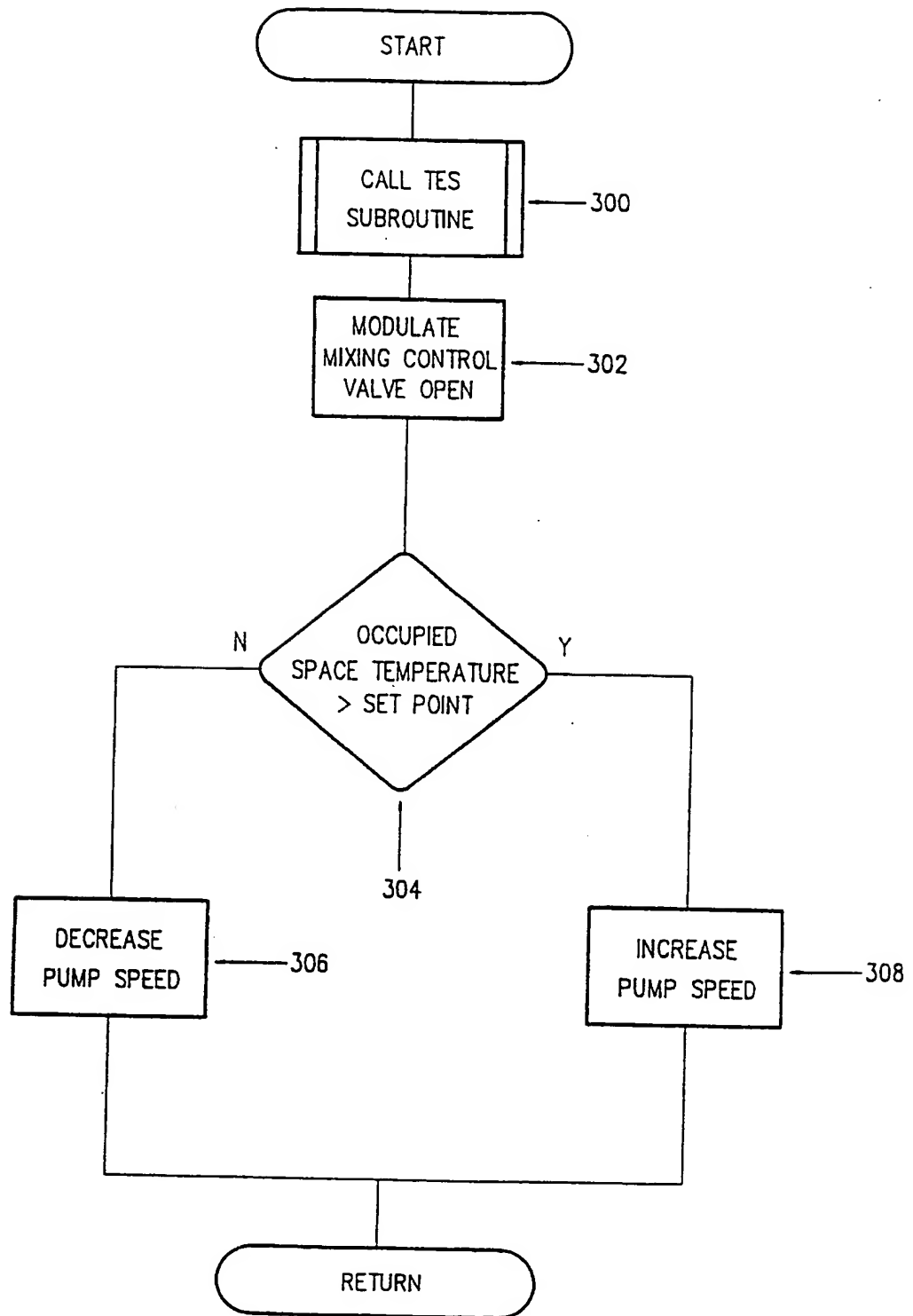


FIG -6

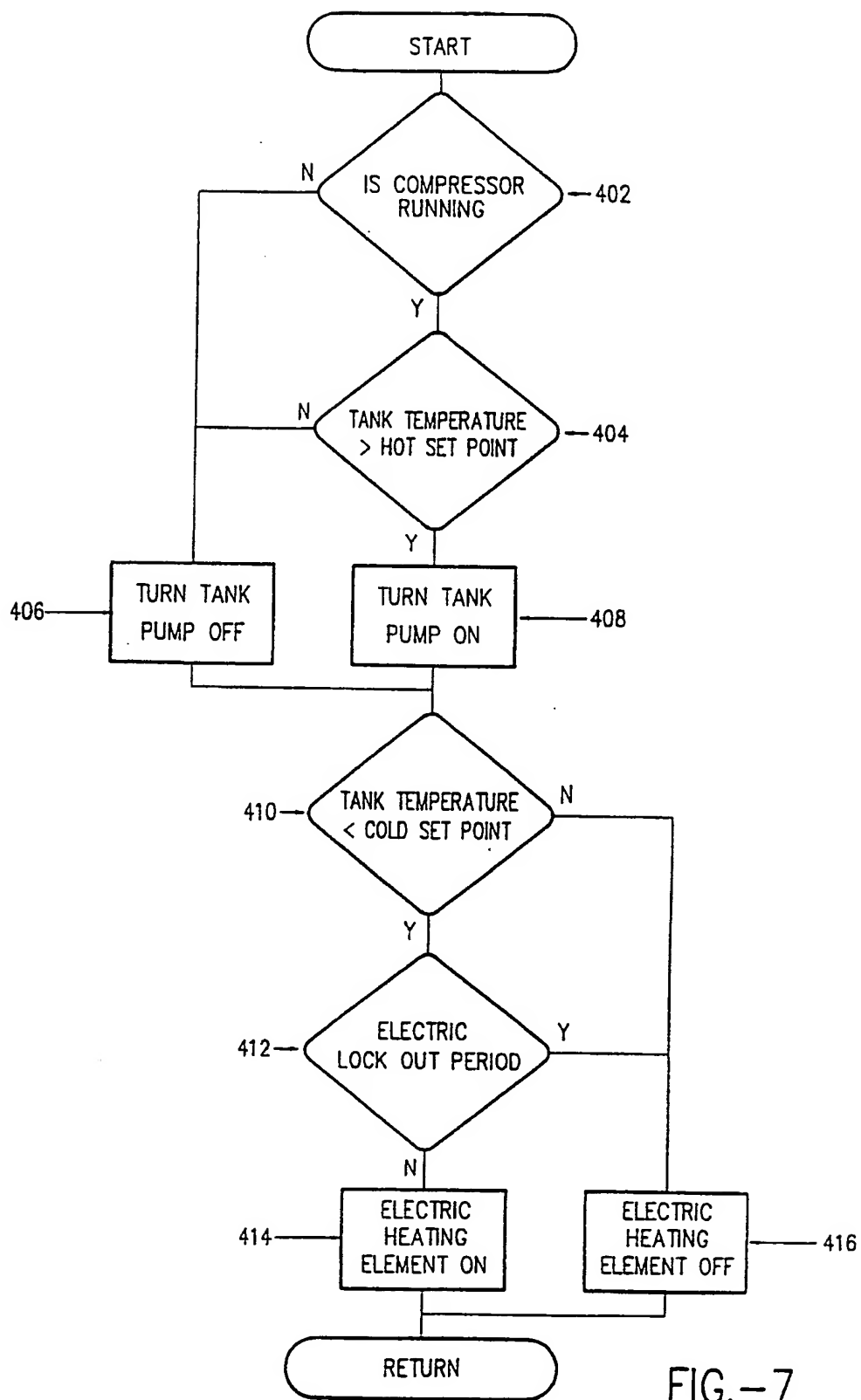
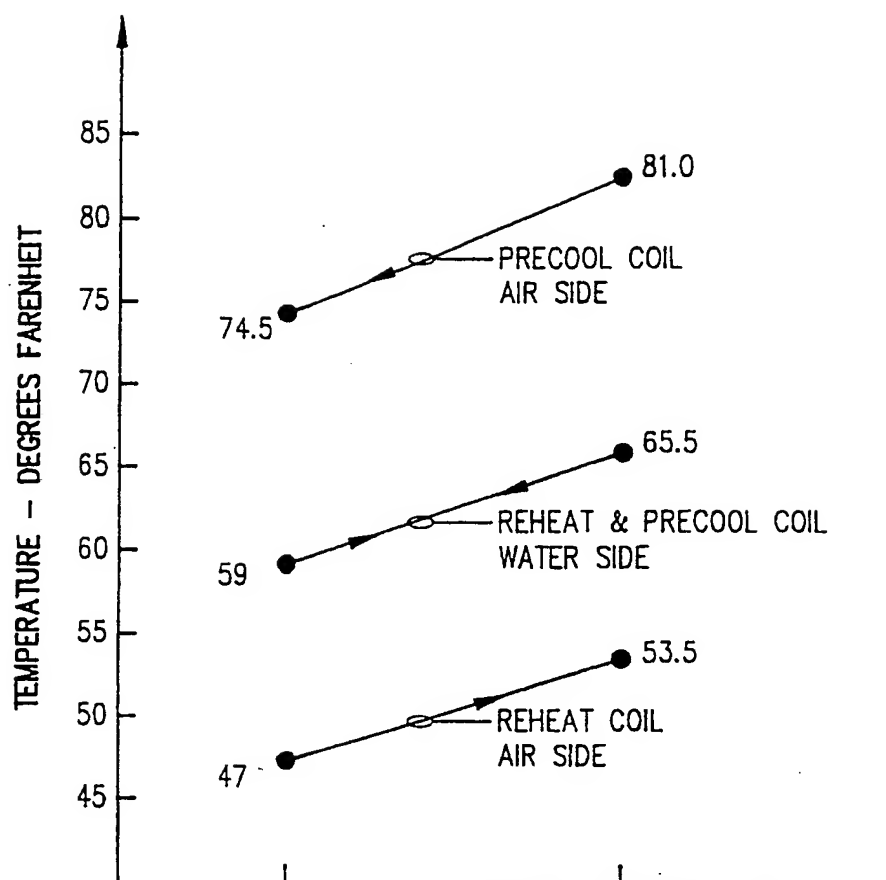
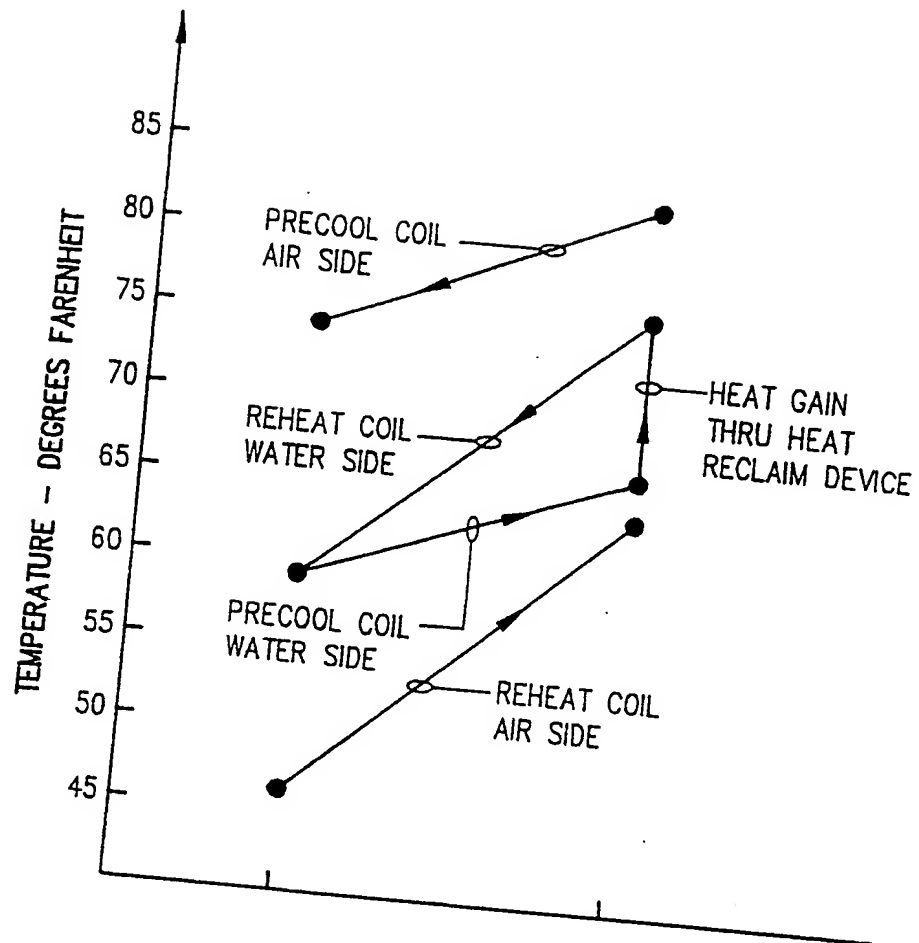
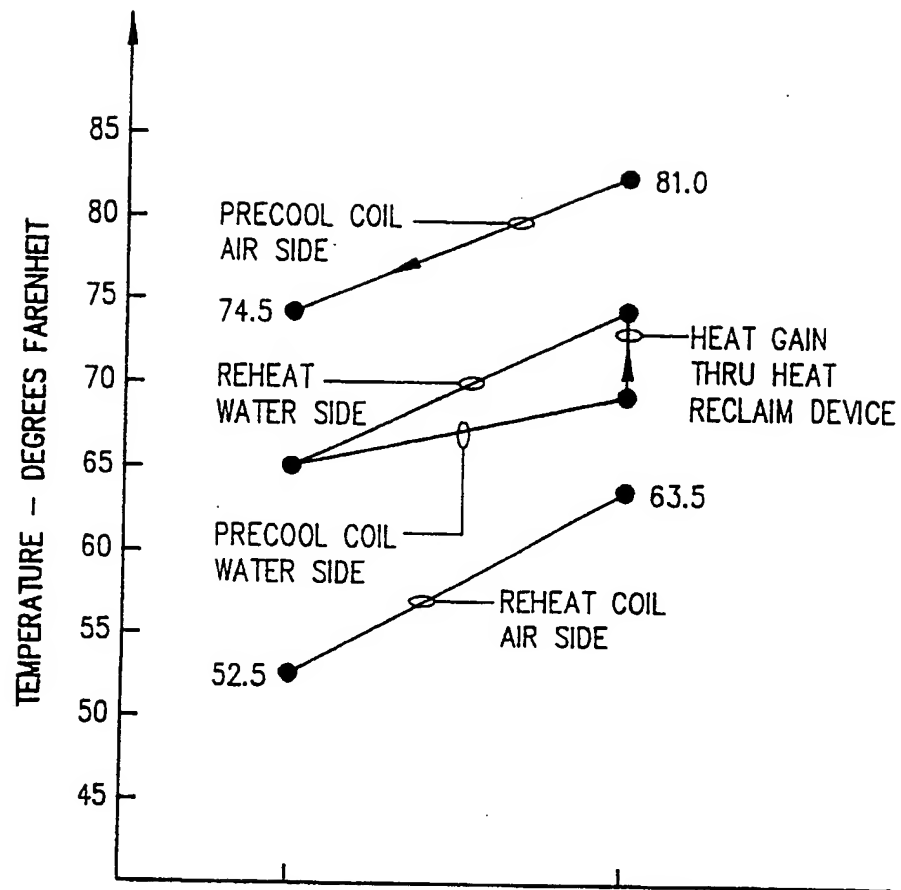


FIG.-7

FIG.-8

FIG.-9

FIG.-10

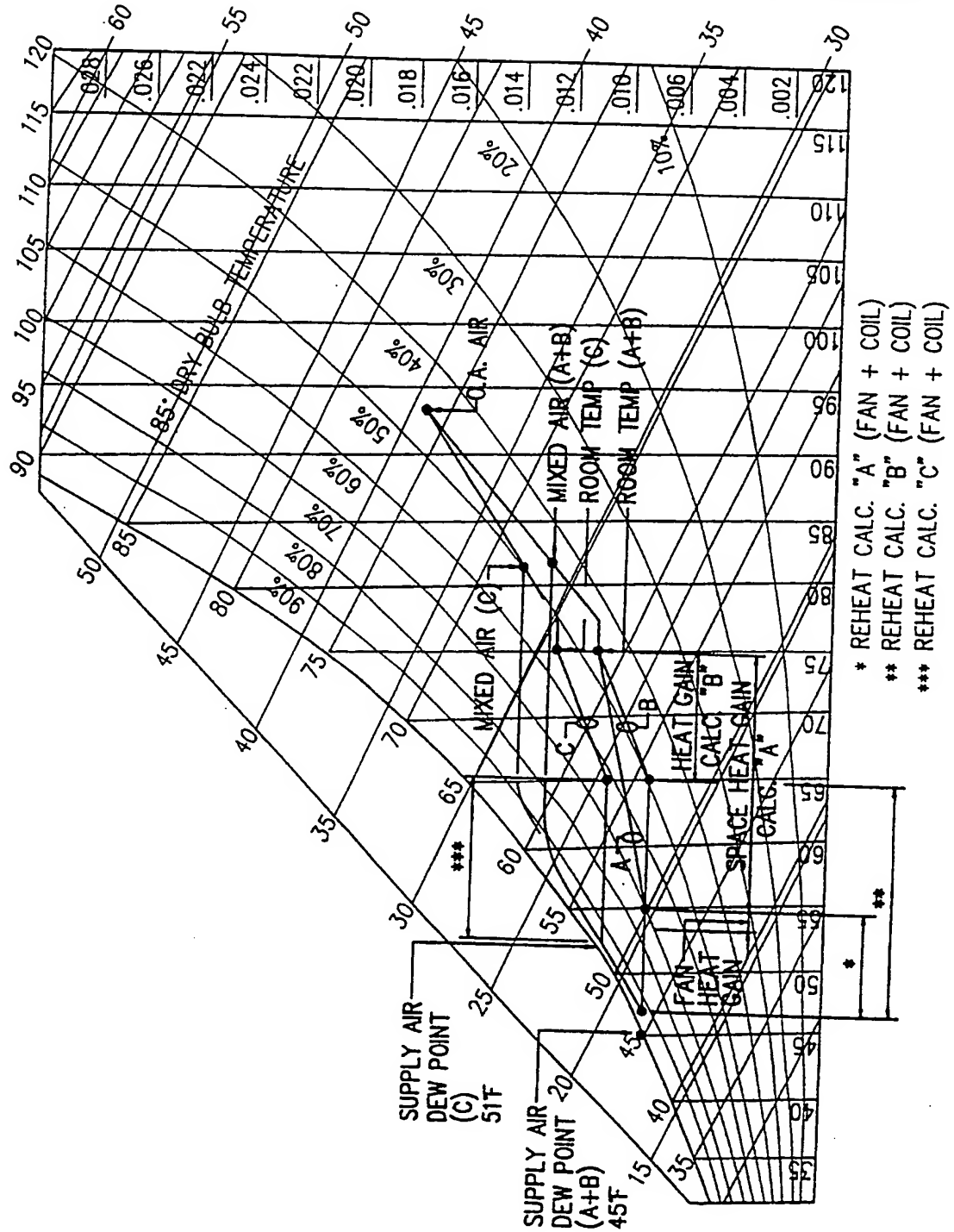
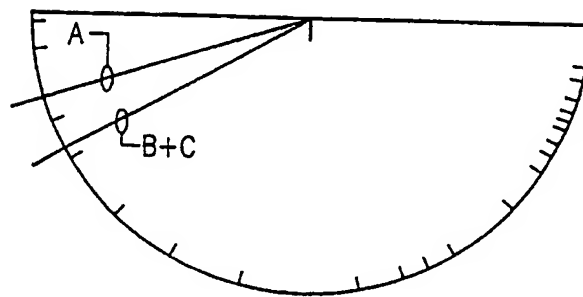


FIG.-11A

14 / 14

SEA LEVEL PSYCHROMETRIC CHART



PROTRACTOR
 $\frac{\text{SENSIBLE HEAT}}{\text{TOTAL HEAT}}$

FIG.-11B

INTERNATIONAL SEARCH REPORT

PCT/US92/09818

A. CLASSIFICATION OF SUBJECT MATTER

IPC(5) : F25D 17/06; F25B 29/00

US CL : 62/90, 62/173

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

U.S. : 62/90, 62/173 62/176.5

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US,A, 2,286,605 (Crawford) 16 June 1942 See Fig. 1.	1-16
A	US,A, 2,291,029 (Everetts, Jr.) 28 July 1942 See Fig. 2, 50,130,114.	1-16
A	US,A, 4,271,678 (Liebert) 09 June 1981 See Fig. 1.	1-16

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:	* T later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
* A document defining the general state of the art which is not considered to be part of particular relevance	* X document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
* E earlier document published on or after the international filing date	* Y document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
* L document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)	* G document member of the same patent family
* O document referring to an oral disclosure, use, exhibition or other means	
* P document published prior to the international filing date but later than the priority date claimed	

Date of the actual completion of the international search

14 JANUARY 1993

Date of mailing of the international search report

03 FEB 1993

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